Optimum Design of a Horizontal Condenser

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Abstract

In this study a rigorous constrained thermal hydraulic model has been derived for calculating the economic design of a vapor surface condenser used in power plants and seawater desalination plants which use distillation methods. According to the economical data obtained in the present study, it has been shown that the total condenser cost components: steam generation, brine water pumping, and fixed charge contribution nearly as 83%, 6%, and 11% respectively according to the fuel cost. However the calculated cost is closed to the known ordinary values when the calculations are based on the minimum variable condenser cost [1] (About 55% for steam cost, 10% for pumping cost, and 35% for the fixed cost). As for the effect of some technical parameters, it has been shown that: (i) total condenser cost is nearly independent of the tube diameter (ii) total cost depends weakly on the brine velocity, and more strongly on the brine flow rate. The plant fouling condition is of prime importance and should not exceed the design values; otherwise it can lead to increase of 126 % of the total condenser cost if the fouling factor increased to 0.05 $m^2\ ^oC/W$. From heat transfer point of view, the effect of the different tube material used in desalination, has a slack effect on the total condenser cost (0.055 – 0.02%). In short, the general model derived in this study is capable to deal with any horizontal condenser.

Introduction

The optimum design of engineering equipment (e.g. evaporator, condenser, heat exchanger, etc) usually involves the calculation of the important design variables, which minimize the total cost of given equipment. However, some constraints are usually imposed on the equipment governing equations so as to get logical or acceptable numerical values for the calculated design variables.

Various optimization methods are available, for example LaGrange's multiplier, linear, dynamic, and geometric programming. The choice of a suitable method depends on the mathematical form of the function to be optimized and type of imposed constraints (e.g. equalities and inequalities). Two efficient optimization methods, which are applicable to nonlinear and multivariable cost functions, are the calculus of variation method and geometric programming. In this respect it has been demonstrated that geometric programming may result in a considerable saving of effort in comparison to the calculus method when the degree of difficulty-equals the number of terms in the objective function and constraints minus the number of variables minus one-is zero. But for a degree of difficulty greater than zero, it may demand more effort than optimization by calculus method. It has been shown that the degree of difficulty in the underlying problem is three, which precludes the use of geometric programming [1] as an optimization method.

In the present study accordingly, we use the method of calculus of variation to determine the economic design of a horizontal surface condenser used in conventional and nuclear power plants and in seawater desalination plants which use distillation methods.

Formulation of the Problem

In the present section the problem of a simplified design of a vapor condenser with fixed heat load like a brine heater used in multi stage flash desalination will be studied. Considering a brine heater in which seawater mass flows at a rate of W (kg/s) and to be heated without phase change from T_{bi} to T_{bo} (^oC) by steam condensation. Optimal design of such brine heater involves minimization of the annual total cost, which includes the following three main items:

Fixed charges on the brine heater,

Cost of circulating seawater in the brine heater,

Cost of steam supply.

The optimization of the brine heater annual cost can be handled through the determination of the brine heater cost algorithm.

Brine heater cost algorithm

The algorithm can be determined be superimposing the before mentioned three annual costs.

The fixed charges on the brine heater C_F (\$/y) can be expressed as follows:

$$C_F = C_a C_d A = \pi C_a C_d DLN = a_7 DLN \quad (1)$$

Where C_a , C_d, and A are the cost per unite area, depreciation and maintenance, and condenser tube surface area respectively.

The Cost of circulating seawater brine to the brine heater C_P ($/y) can be determined as a function of the tube friction pressure drop ΔP_t (D'Arcy equation for tube friction), pressure factor B (typically the tubes friction losses 60-70% of brine recycle pump), friction factor f, pump efficiency η_p, (not including the motor efficiency = 0.7-0.85 [2] typically 0.8 [1], mass flow rate W, and power factor P_F = 0.85 [3], as follows:

$$C_{Pu} = \frac{C_E B \Delta P_t W P_F}{\rho \eta} \tag{2}$$

Where

$$\Delta P_t = \frac{4fL}{D_i}\left(\tfrac{1}{2}\rho V^2\right) \& V = \frac{4W}{\pi D_i^2 \rho N} \tag{3}$$

$$f = \frac{0.046}{\mathrm{Re}^{0.2}} \& \mathrm{Re} = \frac{4W}{\pi \mu D_i N} \tag{4}$$

Substituting from Equations (3&4) in Equation (2) and referring inner tube diameter to the outer, the cost of circulating seawater can be written in the form:

$$C_{Pu} = a_6 D^{-4.8} L N^{-1.8} \tag{5}$$

The cost of the available steam C_S ($/y) can be determined through the thermal load Q and worth of supplied steam S.

So, the total brine heater annual cost C_T ($/y) can be expressed as:

$$C_S = C_{fuel} Q S \tag{6}$$

The worth of the supplied steam S can be correlated to the supplied steam condition, which is a function of steam temperature T_S for saturated steam (by direct referring the cost to the variable fuel cost and correlating the given data in [4]) as follows:

$$S = c_1 + c_2 T_S \tag{7}$$

Again, the steam temperature T_S can be calculated from the heat transfer relations between the steam and the seawater heat balance.

The total resistance to heat flow R_T is the summation of resistances through the inside film R_i, [5] internal tubes fouling R_{fi}, wall thickness R_w, external tubes fouling R_{fo}, noncondensible gases R_{nc}, and the out tube side condensate film R_o [5] represented as:

$$R_T = R_i + R_{fi} + R_w + R_{fo} + R_{nc} + R_o \tag{8}$$

Where,

$$R_f = R_{fi} + R_{fo} + R_{nc} = b \tag{9}$$

Where

$$b_1 = 0.0003[5] - 0.0005[6] \left[\frac{m^2 \cdot {}^o K}{W} \right]$$

$$R_i = \frac{1}{h_i}\frac{D}{D_i} = 1 \Big/ \frac{k_m}{D_i}\left(\frac{4W}{\pi D_i N \mu_m}\right)^{0.8}\left(\frac{c_{pm}\mu_m}{k_m}\right)^{1/3} = b_2 D^{1.8} N^{0.8} \tag{10}$$

$$R_W = \frac{D \ln(D/D_i)}{2\pi \cdot k_W} = b_3 D \tag{11}$$

$$R_O = 1 \Big/ 0.725\left(\frac{g\rho^2 k^3 h_{fg}}{\mu D (T_S - T_{WO})}\right)^{\frac{1}{4}} n^{-\frac{1}{6}} \tag{12}$$

$$n = b_4 \sqrt{N} \tag{13}$$

For calculation simplicity considering the layout of the brine heater tubes has a square form (b_4=1), so the heat flux can be represented as:

$$\frac{Q}{A} = \frac{WC_{PM}(T_O - T_I)}{A} = \frac{T_{FI} - T_M}{R_I} = \frac{T_{WI} - T_{FI}}{R_{FI}} = \cdots = \frac{T_S - T_{nc}}{R_O} = \frac{T_S - T_M}{R_T} \tag{14}$$

So that

$$(T_S - T_{WO}) = R_O \frac{WC_{PM}(T_O - T_I)}{A} \tag{15}$$

Substituting for N and (T_S – T_{WO}) in Equations (13 and 15) in Equation (12) gives:

$$R_O = (0.725)^{-4/3}\left(\frac{\pi g \rho^2 k^3 h_{fg}}{\mu \, WC_{PM}(T_O - T_I)}\right)^{-1/3} L^{-1/3} N^{-2/9} = b_5 L^{-1/3} N^{-2/9} \tag{16}$$

From Equation (14)

$$T_S = T_M + R_T \frac{WC_{PM}(T_O - T_I)}{A} \tag{17}$$

Substituting from Equation (8) for R_T in Equation (17), Equation (17) for T_S in Equation (7), and Equation (7) for S in Equation (6), and rearranging on can get:

$$C_S = a_1 + \frac{a_2}{DLN} + \frac{a_3 D^{0.8}}{LN^{0.2}} + \frac{a_4}{LN} + \frac{a_5}{DL^{4/3} N^{11/9}} \tag{18}$$

The total annual cost can be expressed in terms of C_S, C_{Pu} and C_F as:

$$C_T = C_S + C_{Pu} + C_F \tag{19}$$

Substituting from Equations (18, 2, and 1) for C_S, C_{Pu}, and C_F respectively in Equation (19), the total annual cost can be obtained as follows:

$$C_T = a_1 + \frac{a_2}{DLN} + \frac{a_3 D^{0.8}}{LN^{0.2}} + \frac{a_4}{LN} + \frac{a_5}{DL^{4/3} N^{11/9}} + \frac{a_6 L}{D^{4.8} N^{1.8}} + a_7 DLN \quad (20)$$

From technical point of view, this algorithm is subjected to some constraints concerning tube material, diameter, wall thickness, length, erosive velocity, manufacturing processes, ..etc.

The tube diameter and wall thickness used in desalination article are [6,7]:

$$0.015 \le D \le 0.045(m) \quad (21)$$

and 0.9 up to 1.2 [mm] respectively.

The limiting brine velocity [7] is:

1.8-3 [m/s] for Cu-Ni 90/10,

2-4 for Cu-Ni 70/30

1.5-2.5 for Aluminum Brass (polyphosphate brine chemical treatment plants).

The number of brine heater tubes can be expressed in terms of tube diameter. So, from Equation (3):

$$N = \frac{4W}{\pi \rho_M V D^2} = \frac{a_8}{D^2} \quad (22)$$

Case Study

In the present study a 144 (t/day) MSF plant at Brindisi' power station (Italy) [8] with the following data is taken as working example:

W = 161 (kg/s), seawater concentration = 36,000 (ppm), TBT = 100 (^{o}C), and tubes material Cu-Ni 90/10 39.63 (W/m.^{o}K), thermal conductivity). Take the wall thickness = 1 (mm), diameter ratio = 1.05569, the brine input/output temperature = 91/96 (^{o}C), and the average outside film temperature = 100(^{o}C).

Brine properties [9]

ρ = 988 (Kg/m^3), C_{PM} = 4111.25 (J/kg.K),

k_M = 0.675 (W/m.^{o}K), μ_M = 0.000326 (kg/m.s)

Outside film properties

ρ = 958 (Kg/m^3), C_p = 42195 (J/kg.K), h_{fg} = 2256.9 (KJ/kg)

k = 0.681 (W/m.K), μ = 0.000279 (kg/m.s)

The corresponding constants of the brine heater cost algorithm in equation [20] are:

$a_1 = 229.74913 * C_{fuel} * W$ $a_2 = 7.1789666 * C_{fuel} * W^2$

$a_3 = 822.34417 * C_{fuel} * W^{1.2}$ $a_4 = 2.7201221 * C_{fuel} * W^2$

$a_5 = 1.2774729 * C_{fuel} * W^{(7/3)}$ $a_6 = 3.7344E\text{-}8 * C_{fuel} * W^{2.8}$

$a_7 = 29.845130 * C_{fuel} * W$ (for $C_a = 54$ \$/m^2 [10])

$a_8 = 0.0012887 * W / V_{cr}$

Algorithm Optimization

The method of LaGrange's multipliers is useful and powerful and could very well be the best choice for many practical optimization problems. The brine heater cost algorithm to which LaGrange's multipliers optimization directly applied to the total annual cost represented by Equation (20) is subjected to the constraint:

$$1.8 \le V \le 3(m/s) \tag{23}$$

The optimal value of total cost algorithm C_T is found by the solution of the following scalar non-linear equations [4]

$$\nabla C_T - \lambda \nabla (V - V_{cr}) = 0 \tag{24}$$

$$V - V_{cr} = 0 \tag{25}$$

The grade operator on C_T and $(V - V_{cr})$ in equation (24), is active on D, L, and N so, the following equations can be obtained by direct differentiation of Equation (24) with respect to D, N, and L respectively in Equation (24) as follows:

$$D: -\frac{a_2}{D^2 LN} + \frac{0.8a_3}{D^{0.2} LN^{0.2}} - \frac{a_5}{D^2 L^{4/3} N^{11/9}} - \frac{4.8a_6}{D^{5.8} LN^{1.8}} + a_7 LN + \frac{2a_8 \lambda}{D^3 N} = 0 \tag{26}$$

$$N: -\frac{a_2}{DLN^2} - \frac{0.2a_3 D^{0.8}}{LN^{1.2}} - \frac{a_4}{LN^2} - \frac{11a_5}{9DL^{4/3} N^{20/9}} - \frac{1.8a_6}{D^{4.8} LN^{2.8}} + a_7 DL + \frac{a_8 \lambda}{D^2 N^2} = 0 \tag{27}$$

$$L: -\frac{a_2}{DL^2 N} - \frac{a_3 D^{0.8}}{L^2 N^{0.2}} - \frac{a_4}{L^2 N} - \frac{4a_5}{3DL^{7/3} N^{11/9}} - \frac{a_6}{D^{4.8} N^{1.8}} + a_7 DN = 0 \tag{28}$$

Multiplying Equations (26,27&28) by DL, N L, and L^2 respectively, subtract Equation (26) two times Equation (27), and substituting for N in Equation (22) resulting in:

$$\frac{a_2 D}{a_8} + \frac{1.2a_3 D^{1.2}}{a_8^{0.2}} + \frac{2a_4 D^2}{a_8} + \frac{13a_5 D^{13/9}}{9a_8^{11/9} L^{1/3}} - \frac{1.2a_6 L^2}{a_8^{1.8} D^{1.2}} + a_7 a_8 L^2 = 0 \tag{29}$$

$$-\frac{a_2 D}{a_8} - \frac{a_3 D^{1.2}}{a_8^{0.2}} - \frac{a_4 D^2}{a_8} - \frac{4a_5 D^{13/9}}{3a_8^{11/9} L^{1/3}} + \frac{a_6 L^2}{a_8^{1.8} D^{1.2}} + a_7 a_8 L^2 = 0 \tag{30}$$

From the above two equations, one can obtain the following analytical expression for L as:

$$L = \left(\frac{-\frac{a_2}{a_8} D^2 + \frac{1.4 a_3 D^{2.2}}{a_8^{0.2}} + \frac{11 a_4 D^3}{a_8}}{-\frac{a_6}{a_8^{1.8}} D^{0.2} + a_7 a_8} \right)^{1/2} \tag{31}$$

A transcendental equation in D can be obtained by substituting for L either in Equation (29) or (30) and solving for D. It should be noted that the solution for D must be subjected to the constraint in Inequality (20). So, applying different values of D and V in Inequalities (21) and (23) respectively, one can get the corresponding values of L, N, A, $\mathbf{C_S}$, $\mathbf{C_P}$, $\mathbf{C_F}$, and $\mathbf{C_T}$ at the different values of C_{fuel}, W, V, and D. The obtained results are represented in Figs. (1, 2, 3, and 4).

Results and Discussion

As mentioned before the model derived for estimating the minimum cost of a horizontal surface condenser and hence the corresponding optimum design variables, as tube diameter, tube length and number of tubes, has been applied to 144 [t/day] MSF plant at Brindisi' power station (Italy) [7]. Fig (1A) shows the variation of total cost and specific tube surface area with the tube diameter at an arbitrary brine velocity of 1.9 (m/s), brine mass flow rate of 161 (kg/s) and fuel cost of 1.2 ($/GJ) [11].

It can be seen that there is a weak dependence of the total tube cost and the specific condenser area on the tube diameter. Fig (1B) shows the dependence of both tube length and number of condenser tubes on tube diameter. In this figure, as the tube diameter increases, the number of condenser tubes decreases and the tube length increases. This simple figure can be considered as a design chart for calculating the tube diameter, length, and number of tubes of a horizontal condenser.

The variation of specific area and total cost with the brine velocity is depicted in Fig (2A). It can be seen that the total condenser cost reaches a minimum value at a brine velocity of about 1.8 (m/s). As the brine velocity increase, the condenser specific area decreases. Fig (2B) shows that both the condenser tubes length and number of tubes decrease with the increase of brine velocity.

Figs (3A&3B) show that the specific area and tube length are independent of the brine mass flow rate, while the total cost and number of tubes vary linearly with it.

Fig (4A) shows that the total condenser cost increases linearly with the fuel cost, while the specific area increases less steeply with it. When comparing this figure with Fig (3A), it can be immediately note that the fuel cost is the key parameter which controls the optimization process. This, because it contributes in both the cost of consumed steam (more than 80% of the total condenser cost) and the pumping of seawater, although the latter is much less than the former. Accordingly, it is very important to form the steam cost component and making use of all methods and means to reduce this cost taking into consideration innovation of new technology.

In comparison with the previous work of Avriel-Wilde [1], they made used of geometric programming optimization in solving the condenser problem. They concluded that the designer or economist may reasonably simplify the problem to a zero degree of difficulty to obtain a solution of the problem. So, further assumptions were made to obtain a solution of the problem. They used the minimum variable condenser cost, ignoring the fixed term in the condenser algorithm cost.

However, for comparison purpose, the following results can be obtained when comparing the minimum variable cost percentage of the total cost [1] (at D=0.025 m, L=6.9 m, and N=112 tube)

In the present study: C_F%=42.37, C_{Pu}%=3.08, and C_S%=54.55

While in Avriel-Wilde work: C_F%=45.8, C_{Pu}%=6.1, and C_S%=48.1

These results are almost closed to each other with small differences. These differences may be attributed to elimination of the wall thermal resistance in Avriel-Wilde work, and the different electricity and heating prices.

In the present study, a wide condenser algorithm cost was covered. This enables the designer to get acceptable shortcut optimized cost estimation in accordance with demand and market conditions.

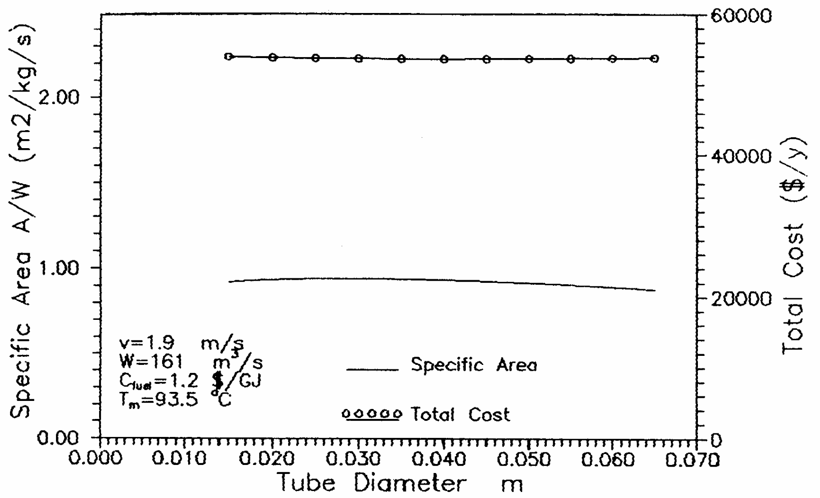
For the purpose for analyzing the effect of individual terms of the cost algorithm, the following analysis of the effect of fouling and tube material is presented. The optimized cost algorithm at W=161 (kg/s), C_{fuel}=1.2 ($/GJ), V=1.9 (m/s), and D=0.019 (m) gives optimized L=8.33 (m) and N=303 tube.

Fouling at plant condition after 30 days normal operation since tube cleaning representing less than 0.27 % of the total condenser cost (at fouling factor R_f= 0.0005 (m^2 ^{o} K/W)). If the fouling factor increased to R_f= 0.05 (m^2 K/W), this can lead to increase to the condenser cost to 126 % of the total condenser cost. So, it should be pointed out that the plant fouling condition is of prime importance and should not exceed the design values.

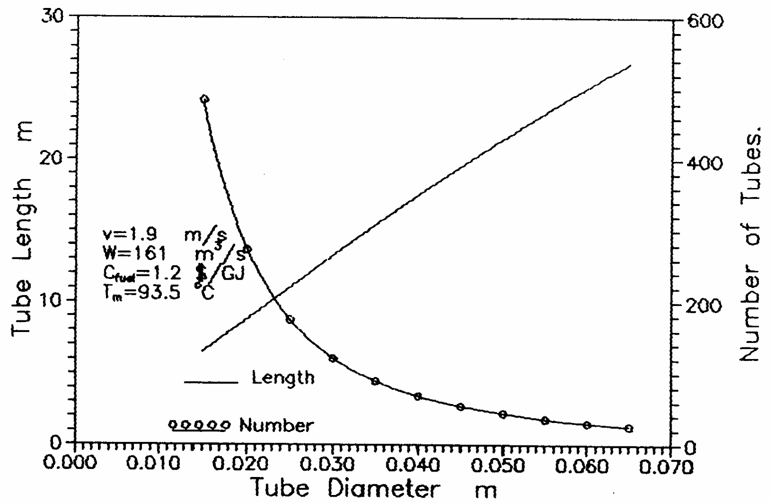
From heat transfer point of view, when studying the effect of the different tube material on the condenser total cost (typically thermal conductivity of materials used in desalination is 13.7 up to 44.7 (W/m.K) [10]), it can be shown that it has a slack effect on the total condenser cost (0.055 – 0.02%).

Nomenclature

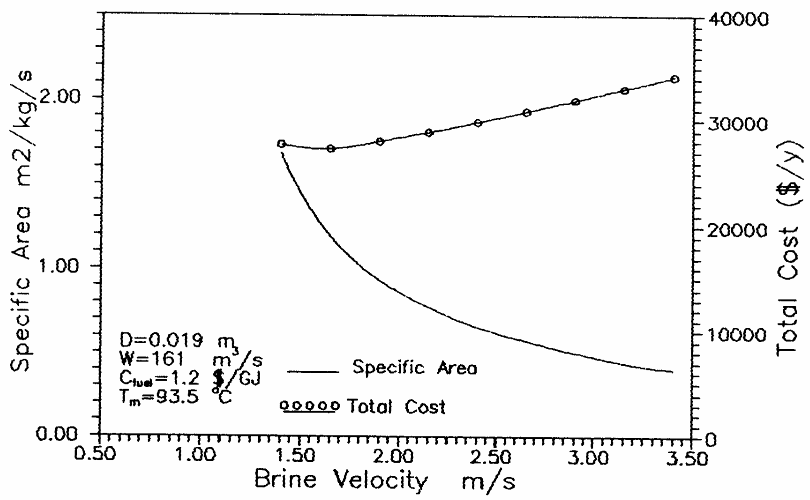
A	Condenser heat transfer area (m^2)
$a_1, a_2, a_3, a_4, a_5, a_6, a_7, a_8$	Constant of Equation (20)
B	Pressure drop factor
b_1	Total fouling Factor ($m^2\ {}^oK/W$)
b_2, b_3, b_4	Constants
c_1, c_2	Constants of Equation (7)
C_a	Cost per unit area ($\$/m^2$)
C_d	Depreciation and maintenance (1/y)
C_E	Cost of electricity ($/kW-hr)
C_F	Cost of fixed charge ($/y)
C_{fuel}	Cost of fuel ($/GJ)
C_p	specific heat (J/kg ${}^oK$)
C_{Pu}	Cost of seawater circulation ($/y)
C_S	Cost of steam ($/y)
C_T	Total annual cost of condenser ($/y)
D, D_i	Outer and inner tube diameters (m)
f	Fanning friction factor
g	Gravitational acceleration (m/s^2)
h	Heat transfer coefficient (W/ $m^2\ {}^oK$)
h_{fg}	Evaporation specific enthalpy (kJ/kg)
k	Heat transfer conduction coefficient (W/ m ${}^oK$)
L	Tube length (m)
N	Number of tubes
P_F	Power factor
Q	Thermal load (GJ/y)



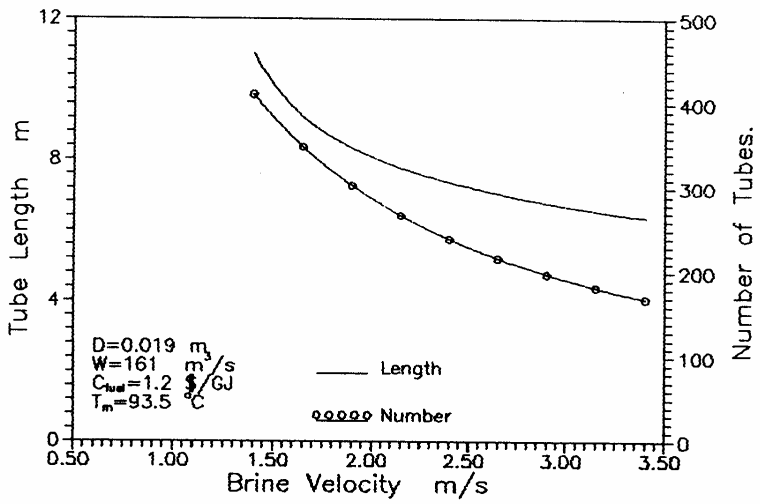
Fig(1A) Variation of total cost and specific area with tube diameter



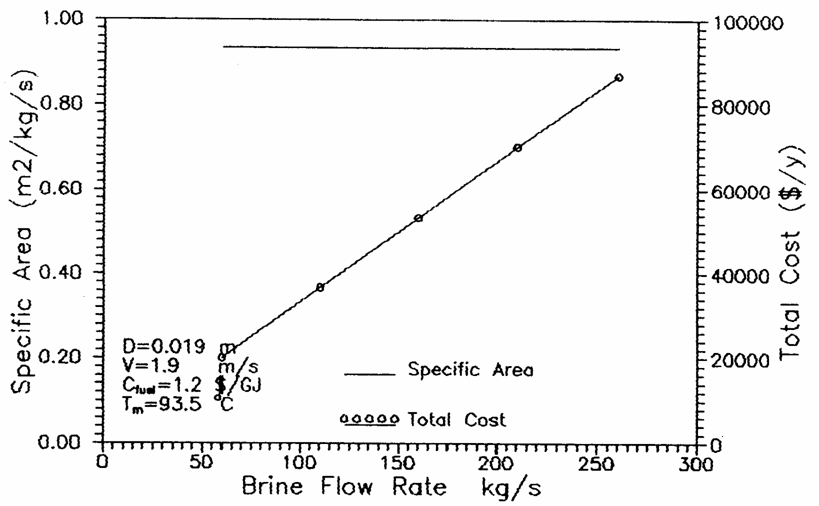
Fig(1B) Variation of length and number of tubes with tube diameter



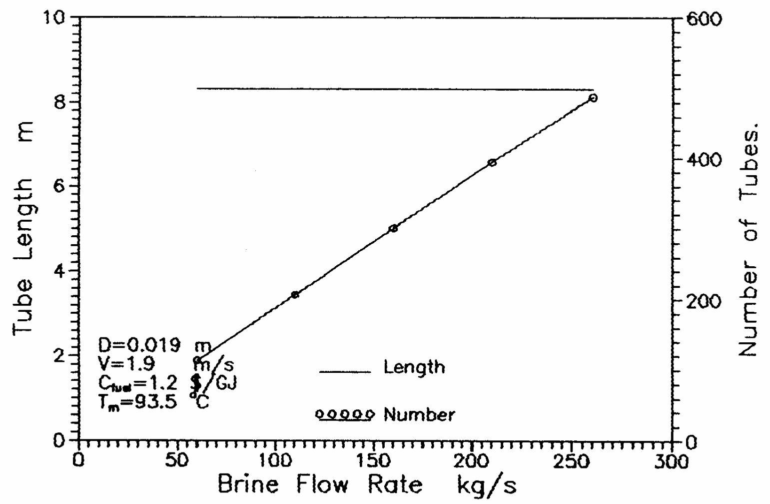
Fig(2A) Variation of total cost and specific area with brine velocity.



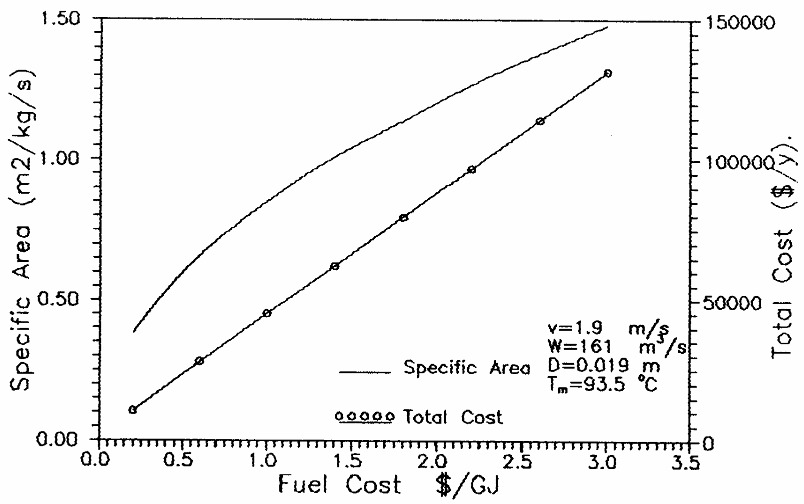
Fig(2B) Variation of length and number of tubes with brine velocity.



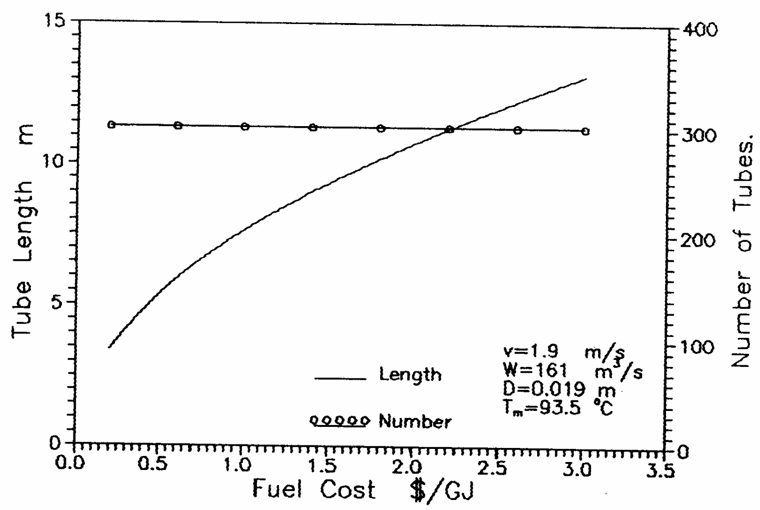
Fig(3A) Variation of total cost and specific area with brine flow rate.



Fig(3B) Variation of length and number of tubes with brine flow rate.



Fig(4A) Variation of total cost and specific area with fuel cost.



Fig(4B) Variation of length and number of tubes with fuel cost.

R	Thermal resistance (m^2 K/W)
S	Steam cost coefficient
T	Temperature (^oC)
V	Brine velocity (m/s)
W	Brine mass flow rate (kg/s)

Greek Letters

ΔP	Pressure drop (N/m^2)
η	Pump efficiency
μ	Dynamic viscosity (kg/ms)
ρ	Density (kg/m^3)

Subscripts

F	Fixed charge
fi	Inner foul
fo	Outer foul
I, i	Inner
M, m	Mean
nc	Noncondensible
O, o	Outer
Pu	Pump
W, w	Wall
WI, wi	Inner wall
WO, wo	Outer wall
S, s	Steam
T	Total

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